AIR FRICTION

ITS ROLE IN TEXTILE SPINDLE POWER CONSUMPTION

A THESIS

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Master of Science in Mechanical Engineering

by

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Approved:

Date Approved by Chairman May 23, 1950

ACKNOWLEDGMENTS

On the completion of this investigation I wish to express my most sincere thanks to the following. To Dr. R. L. Sweigert and The Texas Company for making possible my pursuance of this subject, and to Professor Robert L. Allen, of the Department of Mechanical Engineering, for the conception of the idea expressed by the title of this work.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

SYMBOL DEFINITIONS

Wherever the following symbols appear in these pages, graphs and tables, the definitions thereof are consistent with those stated below.

θ_T Total twist of dynamometer wire, expressed in geometrical degrees.

θ_L Losses of power due to the driving mechanism expressed in geometrical degrees of dynamometer wire twist for the particular speeds tested.

θ_N $= \theta_T - \theta_L$ or net twist of the dynamometer wire - a criterion of spindle power input for the particular speeds tested.

θ_{d1} The difference in θ_T between the yarn weighted bobbin and the wire weighted bobbin.

θ_{d2} The torque of the air friction of the wire weighted bobbin.

θ_{d3} $= \theta_{d1} + \theta_{d2}$

N The r.p.m. of the spindle.

W_T Total power input to spindle in watts.

$= (\theta_N)(N)(0.0000515)$

$= W_A + W_B$

W_A The proportion of W_T consumed by air friction in watts

$= (\theta_{d3})(N)(0.0000515)$

W_B The portion of W_T consumed by spindle bearing friction in watts

$= W_T - W_A$

INTRODUCTION

The role of air friction in textile spindle power consumption is a subject to which little or not attention has been given in the past, whereas a considerable number of studies have been made of the total power consumption of the spindle. Most published articles on the subject of spindle power consumption do not even mention the term "air friction" and the few that do, tacitly leave the reader with the idea that it exists, but is of no considerable consequence.

Tests of power consumption have been made by others where the spindles tested were weighted with an amount of metal equal in weight to the bobbin and the yean which would normally be carried by the same spindle during regular mill operation. It is easy to comprehend that the differences in roughness of surface, in surface area presented to the surrounding air, and in the peripheral speed of the surface area are great, when considered, for example, between 10 ounces of steel and 10 ounces of yarn carried by the same bobbin. The above three differences are all factors, other than the properties the air itself, which determine the difference in power required to overcome the respective air resistances. Obviously the persons responsible for the above tests must have discounted any difference in spindle power consumption due to air resistance.

Results obtained from such means of measuring power consumption would be faulty in proportion to the amount of air friction power actually involved.

Other tests have been made, and data presented, wherein the effects of improved bearing design or improved lubrication or both have been expressed in terms of percent improvement. The improvement is obtained as the equal of total power input reduction in percent. Such results would be misleading if air friction power were of consequence. For example, let us suppose that by some improvement in lubrication we have reduced the power input to a spindle by 30%. If the initial bearing friction were 60% of the initial power input the other 40% being air friction, then the per cent improvement of lubrication, since the air friction at fixed r.p.m., is constant, would be 50% instead of 30%.

$$\frac{30\%}{60\%}(100) = 50\%$$

The main purpose of this investigation was to determine the significance of the air friction, whether it is negligible or whether it is an item worthy of consideration.

A question might be raised as to the importance of finding the air friction at all, total power being the only item worthy of consideration since that is what is paid for. Knowing the air friction isolates the bearing friction which would be of invaluable aid in measuring the progress of improvement in bearing design and lubrication. Furthermore there is no other method for finding the relationships existing between bearing friction, spindle r.p.m. and fullness of bobbin. If air friction is a large factor in spindle power consumption, the knowledge thereof may stimulate the invention of a new spinning process, since the air friction is a product of the high speed spindle and the high speed spindle is the present day means of putting twist into the yarn. If it were not for the twist, the same spindle could wind up the yarn at the same rate at only a few hundred r.p.m. instead of 10,000 r.p.m., for example, with twist. It might therefore prove highly profitable to find a different process for putting twist into the yarn.

The air friction power, as found by this investigation, can actually be greater than the bearing power at 10,000 r.p.m. for the type yarn and spindle used herein.

For those who are also interested in hydrodynamics of lubrication, vibrational effects, etc., a wide variety of subject matter has been included in the bibliography. Those interested in the very modern art of automatic lubrication are directed to a most enlightening, beautifully illustrated article, "Robotized Grease Monkeys," Fortune, July 1949.

APPARATUS

The apparatus used for this experimentation was developed in 1949 by Robert L. Newell, at that time a graduate student at The Georgia Institute of Technology, in conjunction with some technical personnel of the West Point Manufacturing Company.

Figures A and B show two different photographic views of the equipment. The potentiometer, however, was not used for reasons pointed out later. Figure C shows the spindle and bearing components along with the bobbins carrying various quantities of yarn. The lead weights used were added to the empty bobbin as explained later and are now shown.

For a more detailed explanation of this dynamometer, the reader is referred to a thesis, "Power Characteristics of the Textile Spindle," by Robert L. Newell, 1949, available at the Georgia Institute of Technology Library.

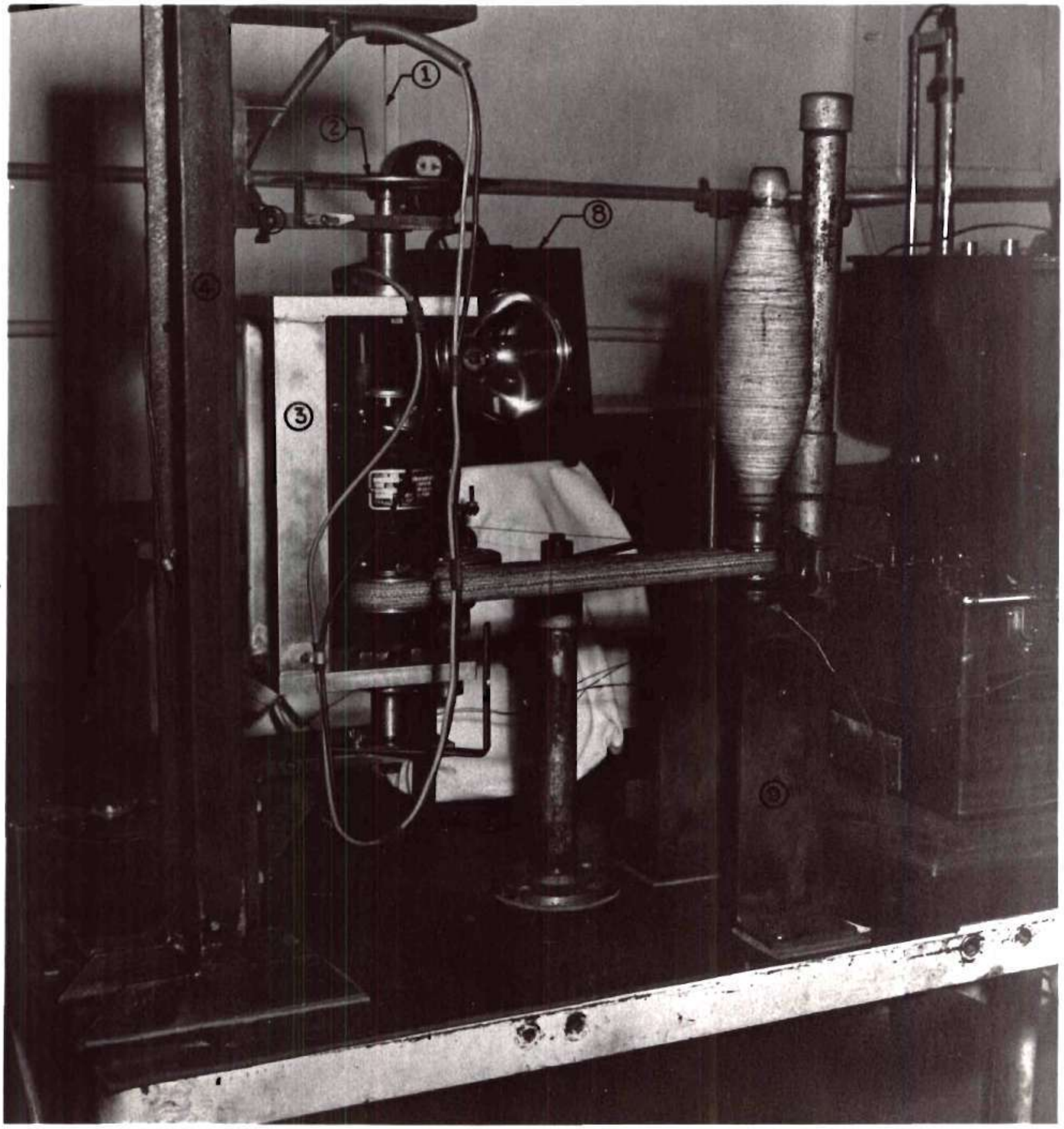


FIGURE A. SINGLE SPINDLE DYNAMOMETER

1. Torsion Wire
2. Dial
3. Motor Support
4. Main Frame
5. Upper Support Bracket
6. Lower Support Bracket
7. Variac
8. Stroboscope
9. Spindle Support

FIGURE B. TOP VIEW OF SINGLE SPINDLE DYNAMOMETER

1. Dial
2. Tension Pulley
3. Tension Weights
4. Potentiometer

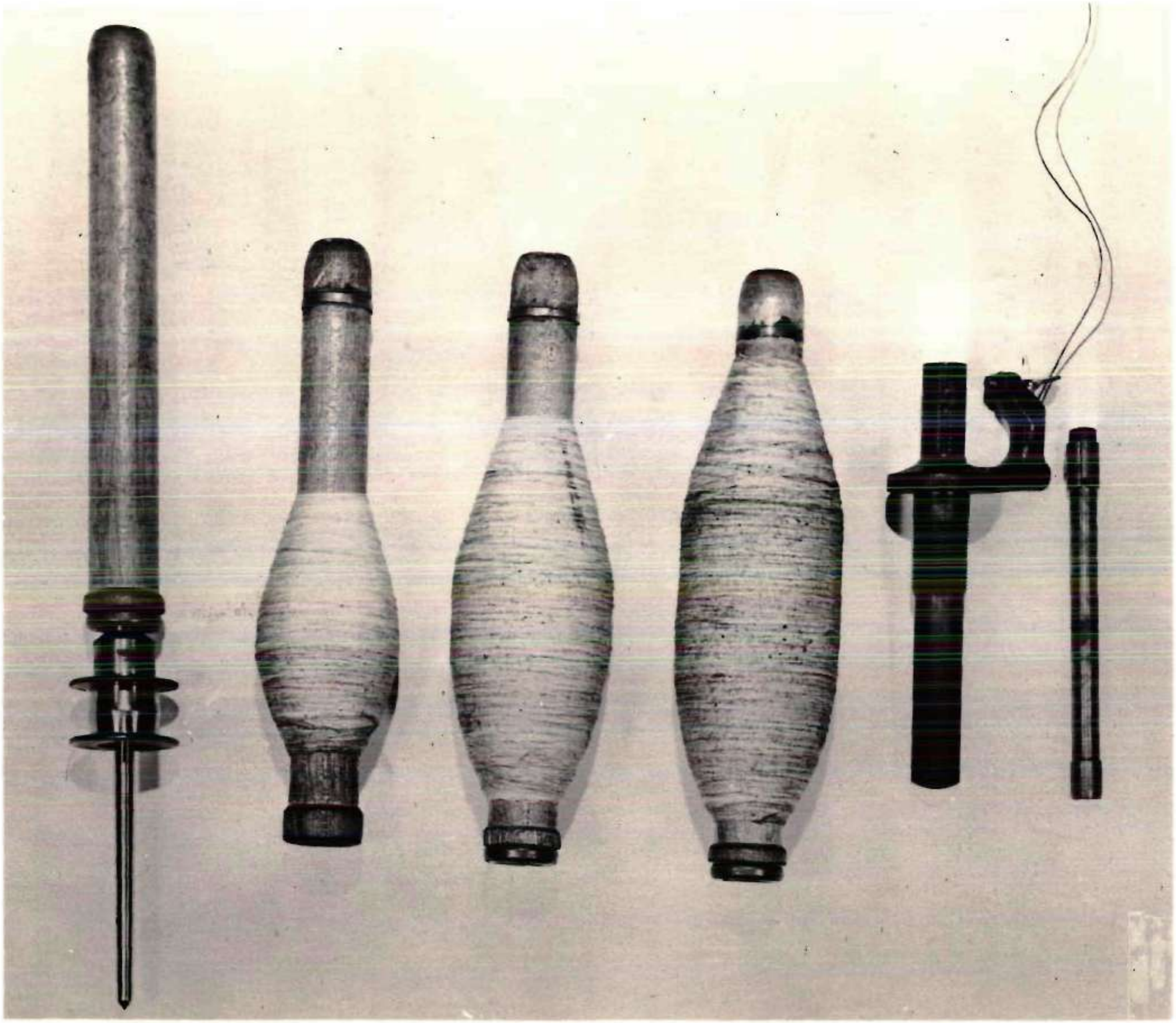


FIGURE C. SPINDLE, BOBBINS, OIL RESERVOIR, AND BOLSTER

METHOD OF EXPERIMENTAL PROCEDURE

The equipment shown and described in the preceeding section was located in the basement of the Mechanical Engineering building in a small, rather tight room.

To obtain the best possible results with this equipment, it was necessary to control the dry-bulb temperature of the room to a greater accuracy than could possibly be achieved by the employment of any ordinary thermostat. The temperature was regulated by manually operating a multitude of high wattage electric light bulbs and was kept within plus or minus $\frac{1}{2}^{\circ}$ F of 75° F.

There are two fairly evident and highly important reasons for desiring such accurate control of the room temperature, and both reasons hinge directly upon the fact that the portion of the power input which is dissipated as bearing friction is disposed of in a steady flow heat transfer process. A temperature gradient is set up from the highest temperature at the film of lubricating oil in the bearing, on through the mounting frame and connecting metallic parts, until it reaches room temperature. Thus, the mechanical power put into the bearing is transformed by friction into heat energy which is in turn disposed of by combined heat transmission, convection, and radiation. In view of the fact that the temperature differences are so small when compared on the absolute scale, we can discount the effects of radiation and for all practical purposes assume that the bearing power consumption is directly proportional to the temperature difference between any particular point in the bearing and room temperature. If the room temperature were not held constant, we could then get an inaccurate comparison of the relative lubricating values of the oils tested and it would furthermore be next to impossible to arrive at a steady flow, or balanced, energy relation since it takes approximately 30 to 60 minutes for the heat flow to level off.

The measurement of operating oil temperatures is a problem which could call for extensive study. The temperatures of the oil inside the bearing will vary considerably according to their closeness to points of frictional contact. This presents the problem of where to measure the oil to get a value for the actual viscosity under operating conditions. The same thermometer measures as much difference as 5° F. between the temperature of the oil in the well, and when strapped onto the outside of the bearing, the latter being the higher. A thermocouple inserted in the oil well close to points of friction gives the highest readings, but the wires in the oil stream no doubt set up some disturbances in the normal flow of the oil. Since this whole study was made primarily for the purpose of isolating the components of power consumption and finding the relative effects of various oils on bearing power, and in view of the fact that the viscosity indexes of all oils tested are nearly straight, parallel lines, the measurement of bearing temperature at any point, when resolved to temperature difference between it and the room air, will serve as a criterion of bearing power consumption which is all that is necessary. The measurement at any one point is also a standard for the actual temperature at any other point, or average temperature of the oil. Those interested in the actual viscosity indexes of the oils tested will find them in the appendix of a thesis, "Power Characteristics of the Textile Spindle" by Robert Lee Newell in the Georgia Institute of Technology Library.

The temperatures of the oil as measured, and as appear in the data sheets of the appendix herein, were measured with a thermometer strapped onto the lower portion of the outside of the bearing housing and covered with a layer of aluminum foil, several layers of tape, and a second, outside layer of aluminum foil. This proceedure reduced to a minimum the difference between the temperature as read and the temperature on the oil side of the housing.

The particular dynamometer setup used in these tests measures the torque on the motor which can then be converted to power for the particular speed at which the torque was measured. The torque is measured in degrees of twist of the torsion wire. However, there are certain losses of power due to the driving mechanism. It would be convenient to express these losses in terms of θ (angle of torsion wire twist) in order to simplify the calculation of net power input to the spindle. These losses will be denoted by θ_L and the calculation thereof proceeds as follows.

The contributing factors to driving power loss (θ_L) are,

Motor pulley air friction,

Idler pulley air friction,

Idler pulley bearing friction,

Belt friction.

The idler pulley was used as a sort of medium in order to find the total driving losses. This statement will appear more enlightening during the analysis which follows.

In order to clarify some of the reasoning in this presentation, figure 1 is included in the argument.

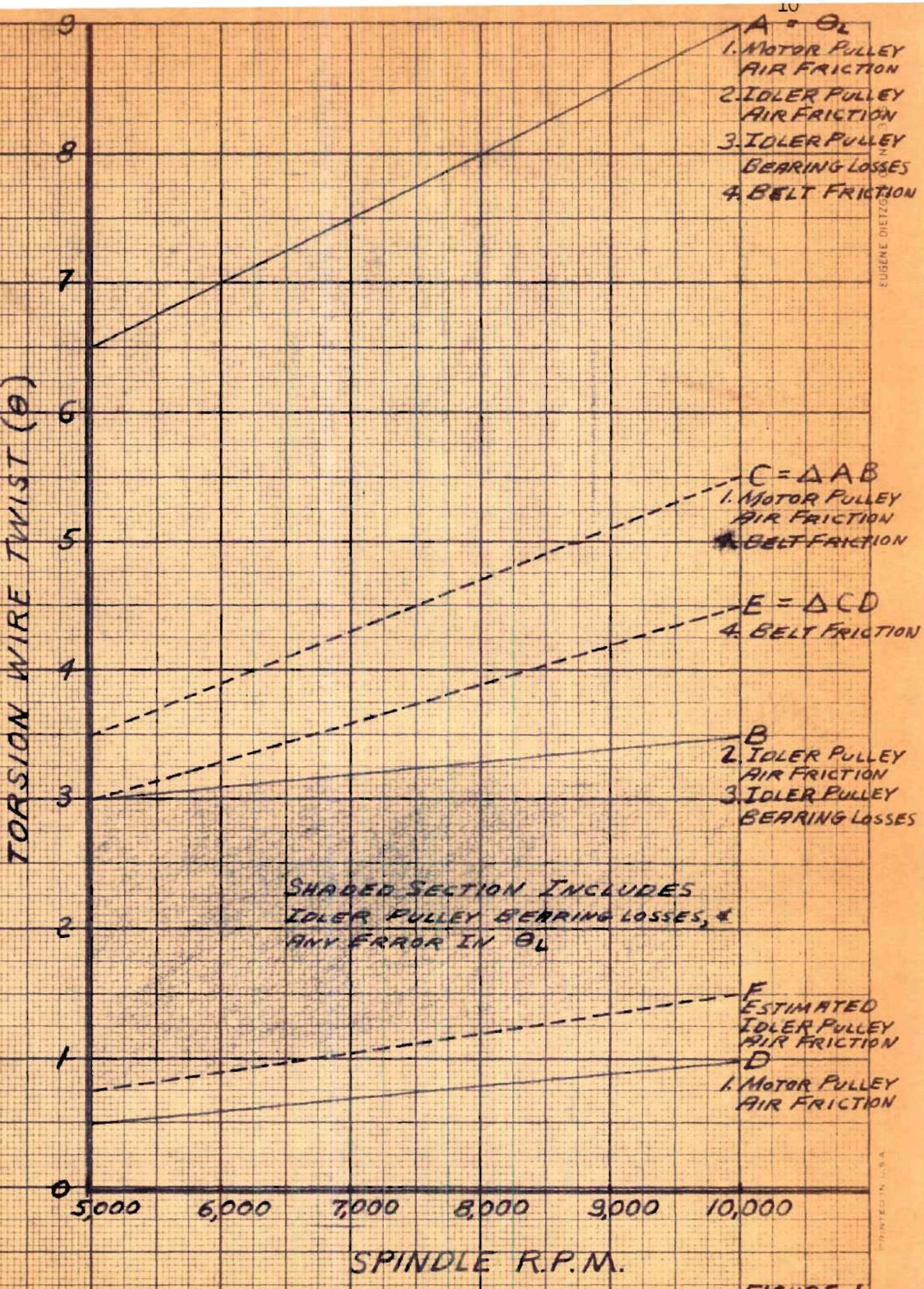

A = θ_L
1. MOTOR PULLEY AIR FRICTION
2. IDLER PULLEY AIR FRICTION
3. IDLER PULLEY BEARING LOSSES
4. BELT FRICTION
C = ΔAB
1. MOTOR PULLEY AIR FRICTION
4. BELT FRICTION
E = ΔCD
4. BELT FRICTION
B
2. IDLER PULLEY AIR FRICTION
3. IDLER PULLEY BEARING LOSSES
SHADED SECTION INCLUDES IDLER PULLEY BEARING LOSSES, & ANY ERROR IN θ_L
F
ESTIMATED IDLER PULLEY AIR FRICTION
D
1. MOTOR PULLEY AIR FRICTION
TORSION WIRE TWIST (θ)
0
1
2
3
4
5
6
7
8
9
5,000
6,000
7,000
8,000
9,000
10,000
SPINDLE R.P.M.

FIGURE 1

The driving belt was disconnected from the spindle and was used to drive the idler pulley at the belt speeds corresponding to the spindle speeds used for this research. The belt tension was adjusted as close as possible to the belt tension when driving the spindle, and the readings, appearing as curve A in figure 1, were taken. This curve, then represents θ_L. However, it was considered that the idler pulley, when running during normal testing, runs against the belt which increases the belt tension somewhat and may create an erroneous effect. The idler pulley was weighted to run against the belt with only enough pressure to prevent noticeable slippage. The possible error is approximated to a very good degree as follows.

A loaded spindle was operated at each of the various speeds required for this research, until equilibrium conditions had been attained and the differences in θ_T were recorded between the system with and without the idler pulley for each speed tested. These differences appear as curve B, figure 1. Curve B, then includes idler pulley air friction, idler pulley bearing friction and losses due to increased belt tension. Determination of the magnitude of the latter is the immediate object of figure 1 and this analysis.

Curve D, figure 1, represents the motor pulley windage loss. It was determined by running the motor, at the various speeds required to drive the spindle for the speeds tested, without the belt attached.

From the relative surfaces of the idler pulley and the motor pulley, a conservative estimate of idler pulley air friction is 150% of the motor pulley air friction. This is graphically represented as curve F, figure 1.

Now, recalling that the components of curve B include any error due to increased belt tension, the differences between curves B and F, shown as the shaded area in figure 1, include the bearing friction of the idler pulley and the above mentioned possible error. It is hardly conceivable that the bearing friction of the idler pulley should not be somewhat greater than its air friction, and it now becomes obviousthat any error in assuming θ_L, as measured previously, to be correct is of no consequence, especially when considered in the light of the total readings, θ_T, taken later.

For general interest, it is convenient to plot some additional curves in figure 1. The solid lines, curves A, B, and D are the result of actual experimentation, as explained previously. Curve C shows the differences between curves A and B. There differences obviously indicate belt friction and motor pulley windage. Curve E represents the differences between curves C and D, showing the belt friction only.

The basis of pertinent experimentation has now been founded and may be expressed simply as,

$$\theta_N = \theta_T - \theta_L$$

and $W_T - (\theta_N)(N)(0.0000515)$

where

θ_T is the total degrees of torsion wire twise at the particular speeds tested,

θ_L is the driving loss, expressed in the same units of torque as θ_T,

N is the r.p.m. of the spindle,

and W_T is the total power input to spindle in watts.

The first step towards the determination of air friction power was to find the total power input to the spindle. The variables used for these tests were,

Oil viscosities (S.U.V.) (100° F.)			216.0
			189.0
			102.6
			83.5
			70.8
Weights of Yarn on Spindle	Full	-	8.58 ounces.
68%	Full	-	5.84 ounces.
38.5%	Full	-	3.30 ounces.
Speed of Spindle	5,000 r.p.m.		
	6,000 r.p.m.		
	7,000 r.p.m.		
	8,000 r.p.m.		
	9,000 r.p.m.		
	10,000 r.p.m.		

The results of these tests appear as tabulated data in columns 1 through 4, of tables I through V, in the appendix. Columns headed 5, in these same tables, show the values of θ_L which were found previously. Columns 6 are columns 4 less columns 5, according to

$$\theta_N = \theta_T - \theta_L.$$

Columns 7 show the values of W_T, total power input to spindle, expressed in watts and calculated from the figures in columns 6 and 1 according to

$$W_T = (\theta_N)(N)(0.0000515)$$

Figures 2 through 5 are curves plotted from the above results.

The next step taken in order to determine the air friction power of the spindle follows.

The empty wooden bobbin was taken and wound with solder wire of equal weight to a full load of yarn. This bobbin, then, would weigh the same as a full bobbin of yarn, but would have little more air frictional resistance than the empty bobbin, since the diameter was increased but very little by the 8.58 ounces of solder. The solder wire had to be wound on the bobbin very evenly, in order to assure good balance. The wire was then covered with one layer of spirally wound friction tape in order to hold it on at high speeds. It was soon discovered that, at high speeds, the wire unwound underneath the tape, indicating imminent catastrophe. In order to solve this latter problem, it was only necessary to nail the ends of the wire to the bobbin with a few small wire brads. The loose end of the friction tape covering had to be wired in place in order to prevent unravelling.

Tests were then run to measure the difference in total torque, θ_T, between the weighted bobbin described above and the bobbin full of yarn. It was immediately evident that the weighted bobbin required no more torque than an empty bobbin running at the same speed with the same lubricating oil, indicating, as was suspected, that the weight, within limits, has no effect on the bearing power requirements.

The above differences in θ_T are recorded in column 2 of table VI in the appendix. These differences, when converted to power, show not the total air friction, but only the difference between the yarn and the wire loaded bobbin. In order to find the total air friction it becomes necessary to add the air friction of the wire loaded bobbin to the above difference in each case. The wire loaded air friction was found by assuming the air friction of the bare metal spindle to be zero for all practical purposes, and then measuring the differences in θ_T between the bare spindle and an empty bobbin covered with a taped surface similar to that of the wire weighted bobbin. These latter differences are tabulated in column 3 of table VI and were added to the column 2 differences, the sums being placed in column 4. Similar tests were then made for the 68% full, and the 38.5% full bobbins, the results of which also appear in table VI.

Column 4, table VI, then, represents the various air frictional torques which, when related to their respective speeds, result according to

$$W_A = (\theta_{d3})(N)\ (0.0000515)$$

as the values of air frictional power consumption, column 5, table VI. Values of W_A are also shown as curves by figure 6.

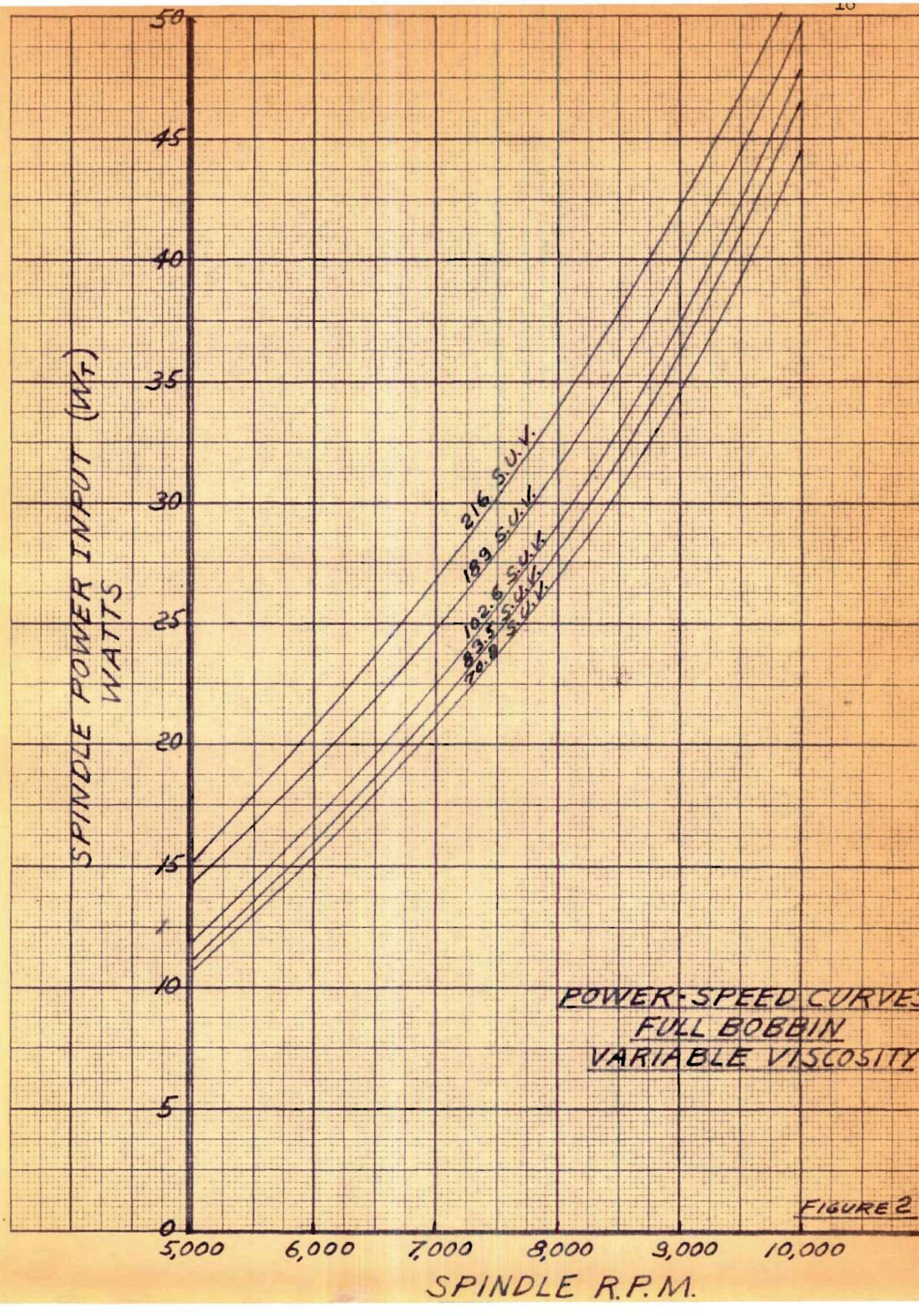
The preceding method for finding air friction need be used on only one sample of oil, since the air friction is obviously independent of bearing friction. The 216 S.U.V. oil was used, however, because of itsgreater stability during operation.

The description of experimental proceedure is now concluded and the following section will show the results in various combinations, with some discussion as to practical application.

RESULTS AND DISCUSSION

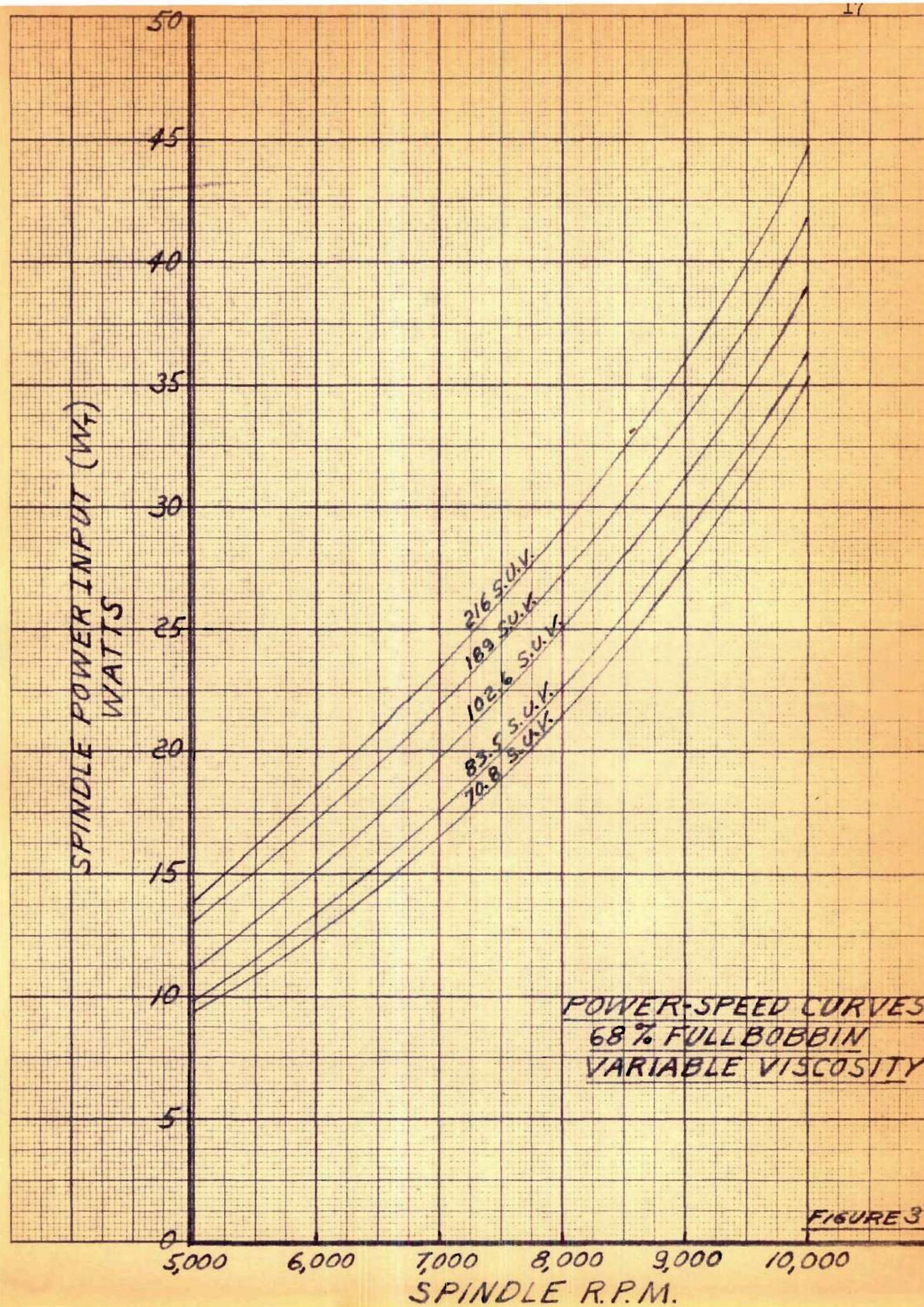
Figures 2, 3, 4, and 5 show the total power input to the spindle, plotted versus spindle speed, for the various oils tested.

The results show approximately 25% greater values of W_T than similar results obtained by Robert L. Newell according to his thesis, "Power Charac-



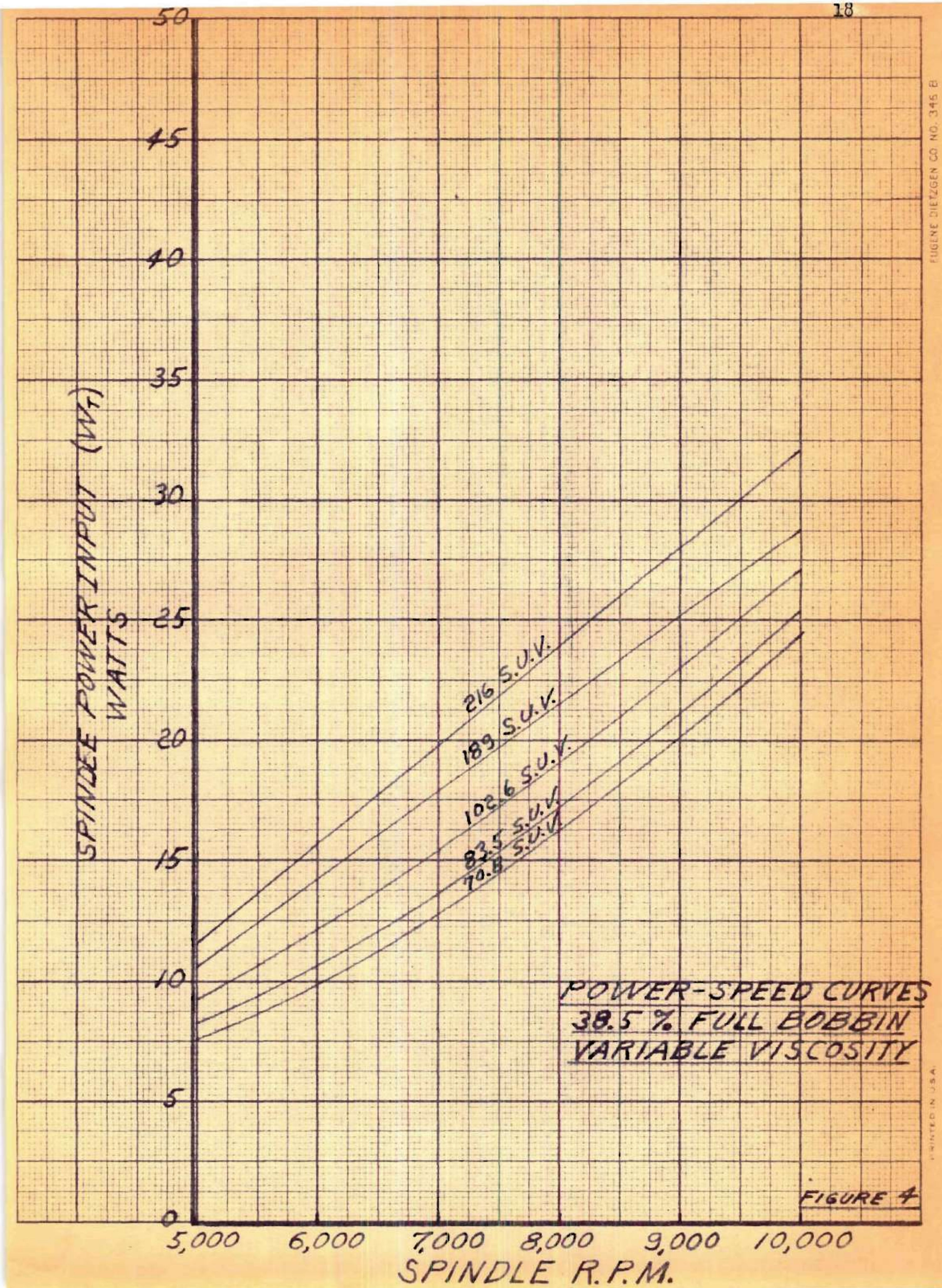
SPINDLE POWER INPUT (W_T)
WATTS
0
5
10
15
20
25
30
35
40
45
50
5,000
6,000
7,000
8,000
9,000
10,000
SPINDLE R.P.M.
216 S.U.V.
189 S.U.V.
102.6 S.U.V.
83.5 S.U.V.
70.8 S.U.V.
POWER-SPEED CURVES
FULL BOBBIN
VARIABLE VISCOSITY

FIGURE 2



SPINDLE POWER INPUT (W_T) WATTS
0
5
10
15
20
25
30
35
40
45
50
5,000
6,000
7,000
8,000
9,000
10,000
SPINDLE R.P.M.
216 S.U.V.
189 S.U.V.
102.6 S.U.V.
83.5 S.U.V.
70.8 S.U.V.
POWER-SPEED CURVES
68% FULL BOBBIN
VARIABLE VISCOSITY

FIGURE 3



50
45
40
35
30
25
20
15
10
5
0
SPINDLE POWER INPUT (W_T)
WATTS
216 S.U.V.
189 S.U.V.
102.6 S.U.V.
83.5 S.U.V.
70.8 S.U.V.
POWER-SPEED CURVES
38.5 % FULL BOBBIN
VARIABLE VISCOSITY
5,000
6,000
7,000
8,000
9,000
10,000
SPINDLE R.P.M.
EUGENE DIETZGEN CO. NO. 345 B
PRINTED IN U.S.A.

FIGURE 4

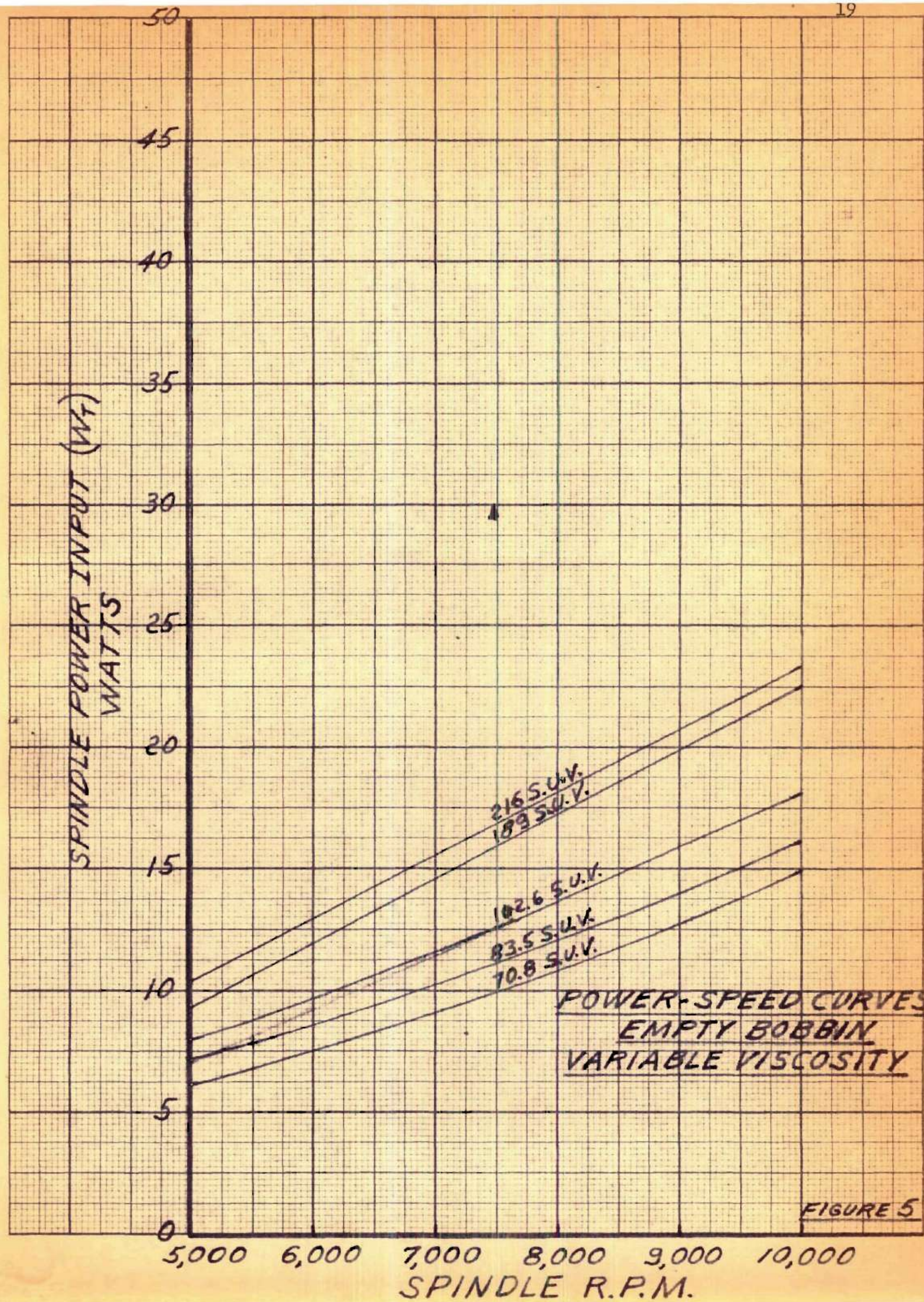

SPINDLE POWER INPUT (W_T)
WATTS
50
45
40
35
30
25
20
15
10
5
0
5,000
6,000
7,000
8,000
9,000
10,000
SPINDLE R.P.M.
216 S.U.V.
189 S.U.V.
102.6 S.U.V.
83.5 S.U.V.
70.8 S.U.V.
POWER-SPEED CURVES
EMPTY BOBBIN
VARIABLE VISCOSITY

FIGURE 5

teristics of the Textile Spindle" which is available in the Georgia Institute of Technology Library. These results, however, can be compared from 5,000 to 9,000 r.p.m. only, since Newell's tests were conducted from 4,000 through 9,000 r.p.m. The tests herein were taken to 10,000 r.p.m. since many present day mills are operating near that level.

Since the tests were run with the same equipment, the calibration of the dynamometer was checked and found consistent, and the oil samples were taken from the same cans, the only explanation that can be offered is that oil samples, due to increased age, may have changed in "oiliness." "Oiliness" is that peculiar characteristic of oil which cannot be defined by other properties thereof. Hunsaker and Rightmire, in "Engineering Applications of Fluid Mechanics," statesthat "oiliness" is the difference between maple syrup and an oil which may be similar to maple syrup in most other respects.

Figure 6 shows the air friction power, plotted versus spindle speed, for the different yarn fullnesses, by weight, of the bobbin. The fact that the two lower curves do not curve upward is probably attributable to the fact that, at the low values involved, the accuracy of calculation, in percent, is lower than at the higher figures. Different forms of bearings or lubricants, of course, would have no effect on these curves.

Figure 7 was plotted from calculation obtained by changing the coordinates of figure 6 to power versus fullness and by graphically integrating under the resultant speed curves to find the average power from empty to full for each particular speed. It is interesting to observe that these average values turned out to be almost exactly equal to the arithmetic mediam averages if taken directly from figure 6. The average power curves

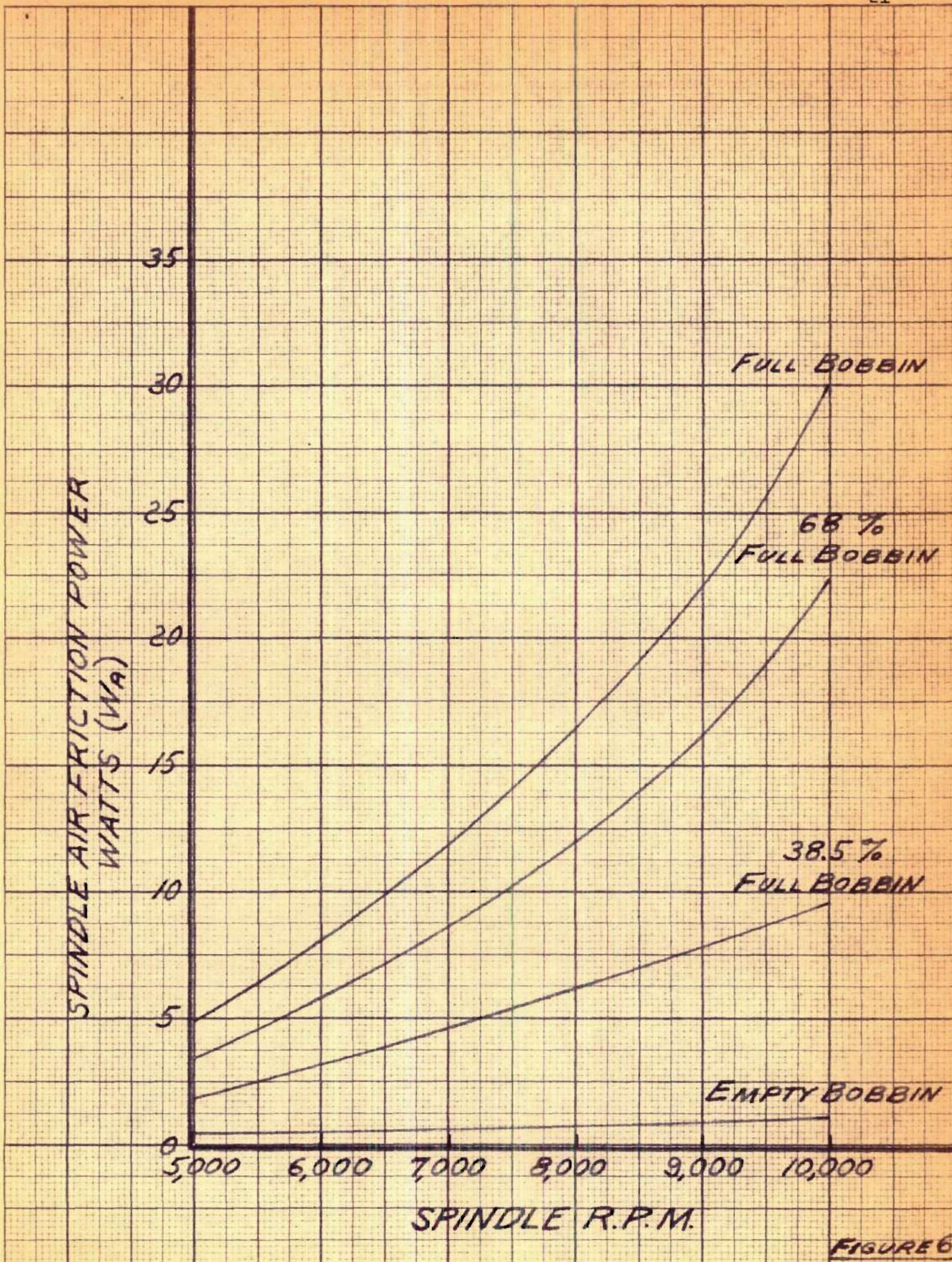

SPINDLE AIR FRICTION POWER
WATTS (W_A)
35
30
25
20
15
10
5
0
5,000
6,000
7,000
8,000
9,000
10,000
SPINDLE R.P.M.
FULL BOBBIN
68 %
FULL BOBBIN
38.5 %
FULL BOBBIN
EMPTY BOBBIN

FIGURE 6

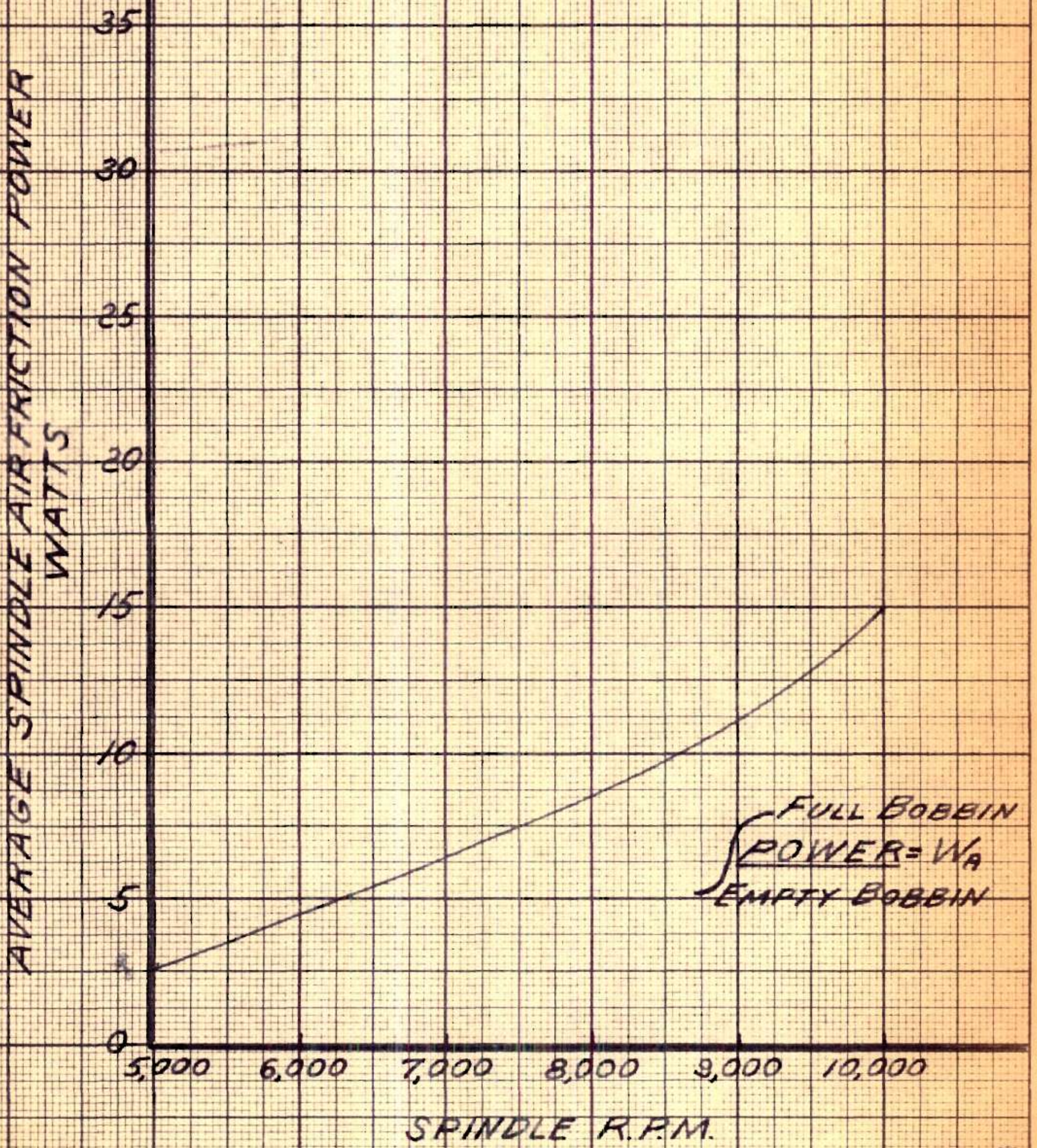

AVERAGE SPINDLE AIR FRICTION POWER
WATTS
35
30
25
20
15
10
5
0
5,000
6,000
7,000
8,000
9,000
10,000
SPINDLE R.P.M.
FULL BOBBIN
POWER = W_A
EMPTY BOBBIN

FIGURE 7

would naturally be applied to economy consideration or other studies requiring average power over a period of time. For example, if a method of spinning were devised which eliminated air friction as a considerable item, the savings obtainable through this elimination could be computed. Other items of expense would, of course, have to be considered also.

Figure 8 shows the bearing power consumption of the spindle, plotted versus spindle speed, for the various oils used. The bearing power W_B, was found according to

$$W_T = W_A + W_B,$$

and the values thereof appear in columns 9 of tables I through IV in the appendix.

Since it was found that bearing power is independent of weight, which, incidentally, is a criteron of full fluid film lubrication, the values appearing in figure 8 are the arithmetic averages of W_B at constant spindle speed for each oil tested.

Such curves should be of great value in determining amounts of improvement in bearing design and lubrication, and also in indicating how much power remains to be saved by further attempts at improvement along these lines.

Figure 9 shows average power input to spindle, plotted versus spindle speeds, for the various oils tested. They are merely the values in figure 7 added to those of figure 8.

These curves are applicable for determining power costs, and show the relative savings to be expected by the use of lower viscosity oils. They are, therefore, probably of most interest to the mill operator.

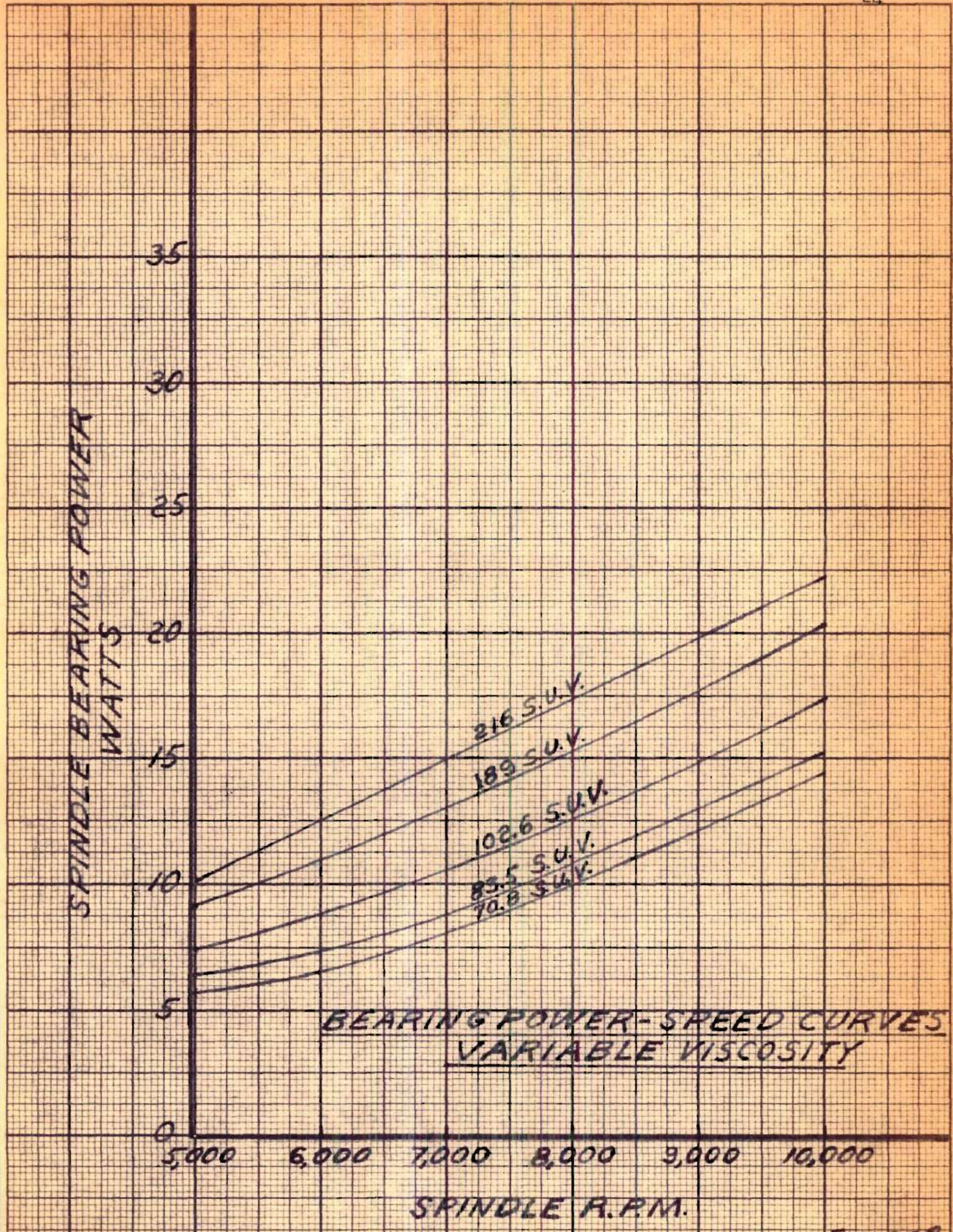

SPINDLE BEARING POWER
WATTS
35
30
25
20
15
10
5
0
216 S.U.V.
189 S.U.V.
102.6 S.U.V.
83.5 S.U.V.
70.8 S.U.V.
BEARING POWER - SPEED CURVES
VARIABLE VISCOSITY
5,000
6,000
7,000
8,000
9,000
10,000
SPINDLE R.P.M.

FIGURE 8

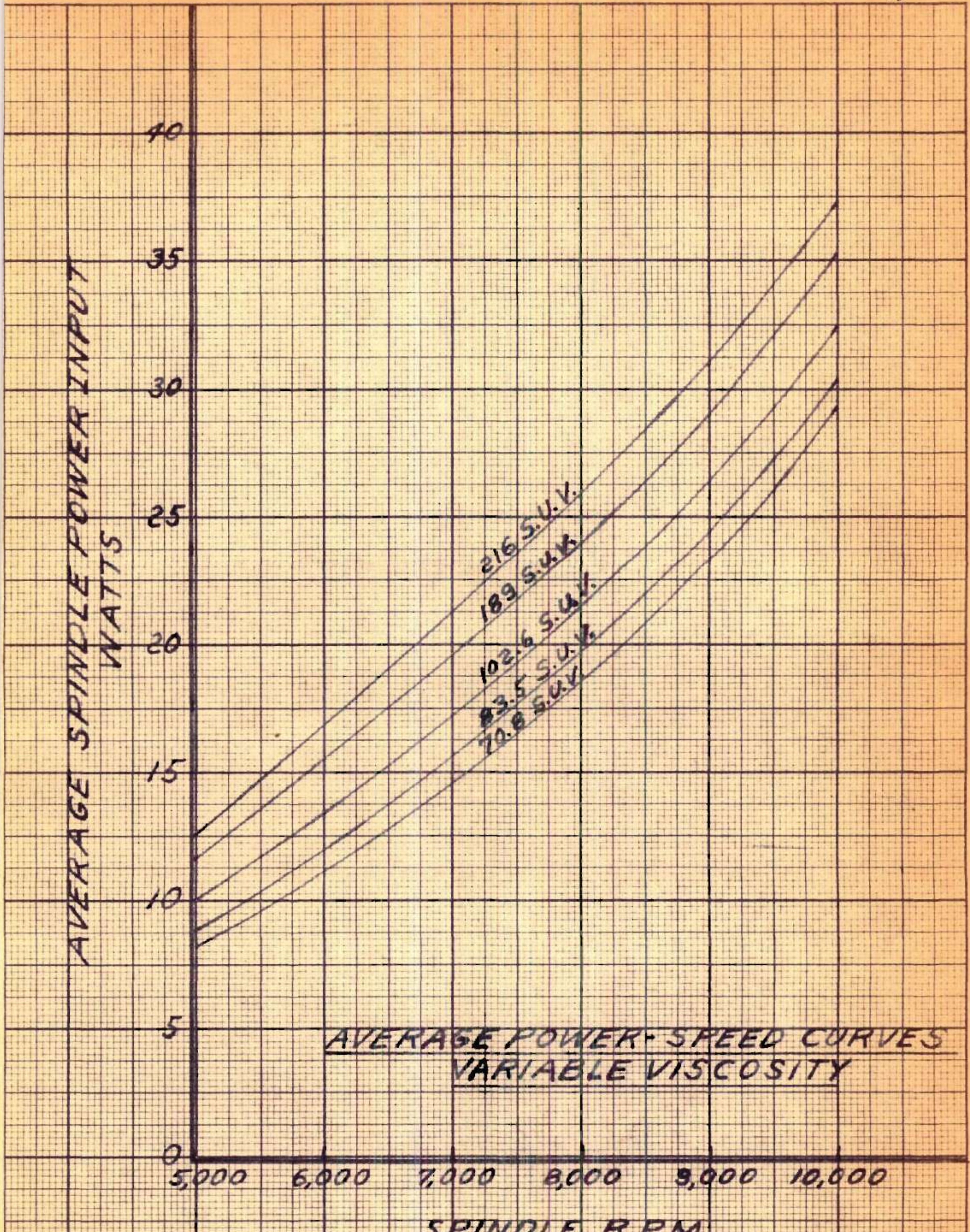

AVERAGE SPINDLE POWER INPUT
WATTS
0
5
10
15
20
25
30
35
40
5,000
6,000
7,000
8,000
9,000
10,000
SPINDLE R.P.M.
216 S.U.V.
189 S.U.V.
102.6 S.U.V.
83.5 S.U.V.
70.6 S.U.V.
AVERAGE POWER-SPEED CURVES
VARIABLE VISCOSITY

FIGURE 9

At this point it may be well to point out that proponents of roller bearings point out great savings due to elimination of the high starting torques prevalent in the plain bearing spindle, which raises greatly the peak electrical load for which mills pay dearly. This assumption is founded on soft ground, since spinning frames are not doffed at the same time and consequently do not start at the same time, but in very staggered intervals. There seems to be room for considerable study on the relative merits of the various bearings on the market today. It should be kept in mind that improvement in bearings can only affect the reduction of bearing power, W_B in this work. The air friction remains constant regardless of any bearing improvement.

Figure 10 showsthe relationships existing between bearing power, air friction power, and total power expressed in % at variable r.p.m. and oil viscosity. These curves show at a glance the comparisons that would otherwise involve the study of other figures in this work, and consequential calculations.

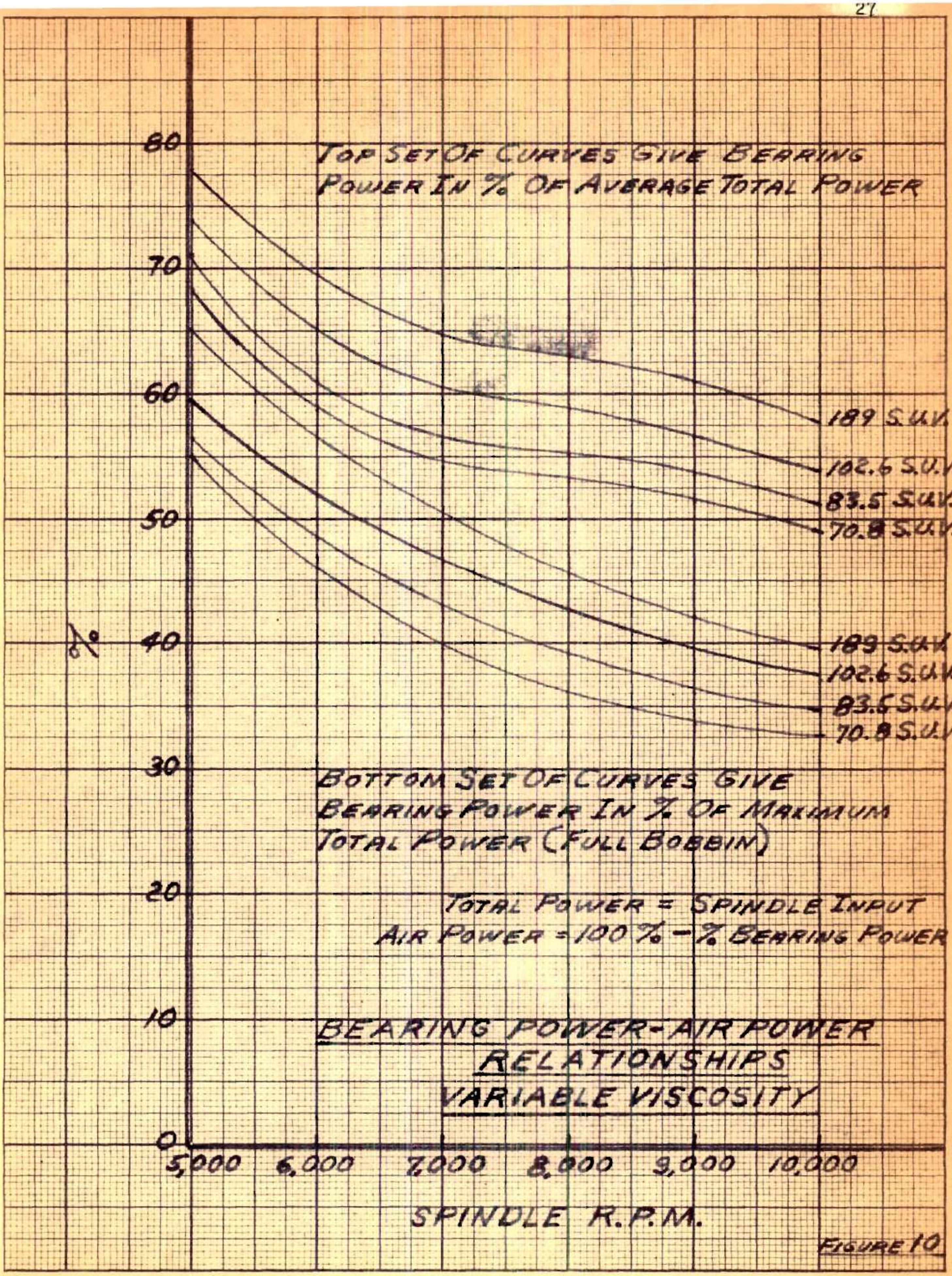
For example, at 10,000 r.p.m. and 70,8 (S.U.V.) oil the bearing power is 49% of the average total power (figure 10, upper set of curves).

Bearing Power 49% x 29.4 = 14.4 watts

Air Friction Power 51% x 29.4 = 15 watts

The figure 29.4 comesfrom figure 9. The answers 14.4 and 15 could be read directly from figures 8 and 7 respectively.

The upper set of curves are relative to power costs, while the lower set involve bearing power in % of maximum total power (full bobbin) and apply



BEARING POWER-AIR POWER RELATIONSHIPS VARIABLE VISCOSITY

FIGURE 10

to installed horsepower calculations. The item of other power losses in the spinning frame, however small, must, of course, also be considered.

It is interesting to note that A. N. Sheldon and J. J. Blake, in "Interrelation of Ring, Bobbin, and Spindle Speed Studied," <u>Textile World</u>, Vol. 95, No. 1, pp. 107, 1945, propose a formula stating that spinning frame power varies as the speed of the spindle to the power 1.808. The authenticity of so many significant figures can honestly be questioned, noting that this combined figure is the result of combining the similar figure for the air friction power, the bearing power, and other parts of the frame, all of which will vary independently of one another depending on every possible variable in the spinning process. Yet, referring to figure 9 herein, the average power requirement for the various oils used, in going from 9,000 to 10,000 r.p.m., varies approximately as the speed to the 1.95 power. Now, considering that the addition of other spinning frame power losses will reduce this figure somewhat because of the minor effect of windage thereon (the windage increases more rapidly with increased speeds than the other losses), it is surprising to note the closeness of the figure 9 relationships with the results obtained on the entire frame in the aforementioned article.

ECONOMY ANALYSIS OF POWER CONSUMPTION

In order to compare the practicability or usefulness of any two or more alternate methods of accomplishing an objective, it is necessary to compare them in the light of economy, which, to any industrial enterprise, is the prime consideration for the selection of means to a specific end.

In the field of textile spindle lubrication, the advantages of various bearings and lubrication oils are usually related directly into kilowatt-hours of annual saving, which, when multiplied by the current cost of power, gives the yearly saving figure in dollars and cents. Such a figure is misleading, and always considerably lower than the actual savings obtainable.

For purposes of illustration, let us assume a typical large mill with 100,000 spindles. We shall further assume that this particular mill is using a spindle oil of variety A, for example. From curves similar to figures 8 or 9 herein, we find that for all the various spindle speeds and types of spinning involved, we can save an average of 5 watts per spindle by switching to oil variety B.

Assuming a power rate of 1¢ per kilowatt-hour and 6,000 hours of annual operation, the commonly used method of finding the savings as explained above would be as follows.

(100,000) spindles (6,000) hours = 600,000,000 Annual spindle hours.
(600,000,000) spindle hours (5) Watts/Spindle = 3,000,000 kilowatt-hours.
(3,000,000) kilowatt hours ($0.01) - <u>$30,000.00</u> Annual savings.

However, further considerations are called for. These additional considerations also depend on whether the change is to be made in

1. An existing mill, or in
2. A proposed new mill.

We shall proceed accordingly, analysing the existing mill first.

A major source of power consumption in the spinning process is that of air conditioning. The average modern mill, maintaining constant temperature the year round, requires approximately 0.5 horsepower of air conditioning equipment for every horsepower of spinning frames. The annual power consumption of the air conditioning system varies directly with the maximum total load in the room. Let us assume that the above reduction in power of 5 watts per spindle constituted a 20% reduction of total spinning power input. The spinning power load consists of approximately 80% of the total maximum room heat gain. Therefore, a 20% reduction in spinning power constitutes a 16% reduction in air conditioning power. This reductiom must be broken down accordingly.

Total Spinning Power Input (Initial)

$$= \left(\frac{5}{0.20}\right)\left(\frac{100,000}{1,000}\right) = 2500 \text{ Kilowatts}$$

Total Air Conditioning Power Input (Initial)

$$(2500)(0.5) = 1250 \text{ Kilowatts}$$

Refrigeration Power Input (60% of 1250) = 750 kilowatts

$$(750) \text{ kw } (2,000) \text{ hour} = 1,500,000 \text{ kilowatt hours}$$

Air Handling, Pumping, etc. Power Input (40% of 1250) = 500 kw hours

$$(500) \text{ kw } (6,000) \text{ hours} = 3,000,000 \text{ kilowatt-hours}$$

Total air Conditioning Power Consumption

$$= \frac{1,500,000 + 3,000,000}{(0.85) \text{ motor efficiency}} = 5,300,000 \text{ kilowatt hours}$$

Air Conditioning Power Saving

$$= 16\% \, (5,300,000)(\$0.01) = \$8,475.00 \text{ Annually}$$

Now, adding the above to the previously calculated spinning power savings, the total savings are obtained.

$$\begin{array}{r} \$30,000.00 \\ \underline{8,475.00} \\ \$38,475.00 \end{array} \quad \text{Total Savings for existing mill.}$$

For a proposed new mill, the above power savings will occur also. However, other savings will ensue as follows.

The saving of 500 kilowatts in spinning power input means a saving in total power of

$$\frac{500 \text{ kw}}{(0.85) \text{ motor efficiency}} = 588 \text{ kilowatts}$$

This saving would result in the saving of approximately

$$(588)(\$40.00) = \$23,520.00$$

in electrical utility installation. Assuming 10 year life and a money value of 6% interest, the annual value of this saving is

$$\$23,520.00\ (0.13587) = \$3,195.00$$

Similarly, we can calculate the savings in Air Conditioning installation.

$$\frac{1250}{.85}\ (16\%) = 235 \text{ kilowatts}$$

This is the total saving in installed air conditioning power. At current prices of such systems, this saving amounts to

$$(235 \text{ kw})(\$300/\text{kw}) = \$70,500.00$$

Again using 10 year depreciation with 6% interest, we get an annual saving of

$$(\$70,500.00)(0.13587) = \$9,575.00$$

Thus, the total savings for the proposed new mill, by the reduction of 5 watts per spindle or 20% of spinning power input, amounts to

$38,475.00
3,195.00
9,575.00
$51,245.00 Total annual savings

When comparing this figure with the $30,000.00 saved by spinning power reduction alone, we find that, for every watt-hour saved in spinning power, we have actually saved the price of 1.71 watt-hours. This is well worth considering.

For a smaller mill than the one in this analysis, the relative saving will be slightly greater because of greater unitary costs of equipment.

CONCLUSIONS

The results obtained, in this experimentation, for the values of air friction are for one particular type of yarn only. Every factor which might affect the roughness of the spindle package, such as the nature of the material (cotton, wool, rayon, etc.), the twist of the yarn per inch of length, the tightness of winding, the humidity of the room, etc. will have effect upon the actual air friction involved. It is easily seen that to obtain these values for all possible combinations of variables would involve endless experimentation.

This study, however, has shown air friction to be the major part of spindle power consumption for the particular yarn used at the highest spindle r.p.m. Implication follows that by far greater savings could be effected by the development of a new way of spinning, a way which could put twist into yarn without such high speeds, than any savings induced by improvement of bearings. Bearing friction cannot be eliminated, only reduced. As was previously pointed out, only a few hundred r.p.m. of the spindle are required to wind the yarn at a production rate equivalent to 10,000 r.p.m. with twist.

However, since the development of such a process is apparently far over the horizon, we must make the best use of what we have. A study should be made to determine, as outlined herein, the actual bearing power of other types of bearings. According to the lubricants tested, it seems ill advisable to use lower viscosity oil in the plain type bearing. The lowest viscosity used seemed to increase the audible effects of vibration slightly at the higher speeds. Furthermore, the speed became considerably more unstable, which would result in uneven yarn twist. It is suggested however, that ultra-low viscosity oils be tested in combination with all potential vibration reduction components such as very high gravity and/or extreme pressure additives.

Careful attention should be paid to the sample economy study presented herein. It clearly indicates that any savings obtainable by the use of better lubricants are far greater than they would appear on the surface.

This experimentation has conclusively proven that the bearing power of the type bearing tested does not vary with the weight of yarn on the bobbin, but only with speed and type of lubricant. Air friction, however, increases at a great rate with the increase of package size, due to increased area and increased peripheral speed at constant r.p.m., until, when the bobbin is full, the air friction outweighs the bearing friction by a considerable amount at higher spindle r.p.m.

Since it was shown that weight is of no consequence at normal operating speeds, for the plain type bearings used herein, further tests on lubrication improvement may be run without the yarn package on the spindle, thus reading bearing power directly.

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APPENDIX

TABLE I

216 S.U.V. (100°F.) Viscosity Oil

R.P.M. SPINDLE	AMBIENT AIR °F	OIL TEMPERATURE °F	θ_T °TWIST	θ_L °TWIST	θ_N °TWIST	W_T WATTS	W_A WATTS	W_B WATTS
1	2	3	4	5	6	7	8	9
			FULL BOBBIN					
5,000	$75 \pm \frac{1}{2}$	88.5	65.0	6.5	58.5	15.10	4.90	10.20
6,000	$75 \pm \frac{1}{2}$	91.0	73.5	7.0	66.5	20.55	8.10	12.45
7,000	$75 \pm \frac{1}{2}$	94.0	83.0	7.5	75.5	27.20	12.40	14.80
8,000	$75 \pm \frac{1}{2}$	96.5	90.5	8.0	82.5	34.00	16.70	17.30
9,000	$75 \pm \frac{1}{2}$	100.0	98.5	8.5	90.0	41.70	21.90	19.80
10,000	$75 \pm \frac{1}{2}$	103.0	110.5	9.0	101.5	52.30	29.90	22.40
			68% FULL BOBBIN					
5,000	$75 \pm \frac{1}{2}$	88.5	59.0	6.5	52.5	13.52	3.35	10.17
6,000	$75 \pm \frac{1}{2}$	91.0	66.5	7.0	59.5	18.40	5.93	12.47
7,000	$75 \pm \frac{1}{2}$	94.0	73.0	7.5	65.5	23.60	8.80	14.80
8,000	$75 \pm \frac{1}{2}$	96.5	78.0	8.0	70.0	28.82	11.56	17.26
9,000	$75 \pm \frac{1}{2}$	100.0	86.0	8.5	77.5	35.90	16.12	19.78
10,000	$75 \pm \frac{1}{2}$	103.0	95.0	9.0	86.5	44.60	22.10	22.50
			38.5% FULL BOBBIN					
5,000	$75 \pm \frac{1}{2}$	88.5	52.0	6.5	45.5	11.70	1.55	10.15
6,000	$75 \pm \frac{1}{2}$	91.0	60.5	7.0	53.5	16.52	4.08	12.44
7,000	$75 \pm \frac{1}{2}$	94.0	63.5	7.5	56.0	20.20	5.37	14.83
8,000	$75 \pm \frac{1}{2}$	96.5	65.5	8.0	57.5	23.70	6.42	17.28
9,000	$75 \pm \frac{1}{2}$	100.0	67.5	8.5	59.0	27.40	7.55	19.85
10,000	$75 \pm \frac{1}{2}$	103.0	71.0	9.0	62.0	32.00	9.53	22.47
			EMPTY BOBBIN					
5,000	$75 \pm \frac{1}{2}$	88.5	47.0	6.5	40.5	10.42	0.26	10.16
6,000	$75 \pm \frac{1}{2}$	91.0	48.5	7.0	41.5	12.81	0.37	12.44
7,000	$75 \pm \frac{1}{2}$	94.0	50.0	7.5	42.5	15.32	0.51	14.81
8,000	$75 \pm \frac{1}{2}$	96.5	51.5	8.0	43.5	17.92	0.66	17.26
9,000	$75 \pm \frac{1}{2}$	100.0	53.0	8.5	44.5	20.62	0.83	19.79
10,000	$75 \pm \frac{1}{2}$	103.0	54.5	9.0	45.5	23.44	1.03	22.41

TABLE II

189 S.U.V. (100°F.) Viscosity Oil

R.P.M.	AMBIENT AIR	OIL TEMPERATURE	θ_T	θ_L	θ_N	W_T	W_A	W_B
SPINDLE	°F	°F	°TWIST	°TWIST	°TWIST	WATTS	WATTS	WATTS
1	2	3	4	5	6	7	8	9
			FULL BOBBIN					
5,000	75 ± ½	86.5	62.0	6.5	55.5	14.30	4.90	9.40
6,000	75 ± ½	88.5	69.0	7.0	62.0	19.15	8.10	11.05
7,000	75 ± ½	91.0	75.0	7.5	67.5	24.35	12.40	11.95
8,000	75 ± ½	93.5	83.0	8.0	75.0	30.87	16.70	14.17
9,000	75 ± ½	97.0	94.0	8.5	85.5	39.62	21.90	17.72
10,000	75 ± ½	100.5	105.5	9.0	96.5	49.70	29.90	19.80
			68% FULL BOBBIN					
5,000	75 ± ½	86.5	56.0	6.5	49.5	12.75	3.35	9.40
6,000	75 ± ½	88.5	62.0	7.0	55.0	17.00	5.93	11.07
7,000	75 ± ½	91.0	67.5	7.5	60.0	21.61	8.80	12.81
8,000	75 ± ½	93.5	73.0	8.0	65.0	26.77	11.56	15.21
9,000	75 ± ½	97.0	81.0	8.5	72.5	33.60	16.12	17.48
10,000	75 ± ½	100.5	90.0	9.0	81.0	41.70	22.10	19.60
			38.5% FULL BOBBIN					
5,000	75 ± ½	86.5	47.5	6.5	41.0	10.56	1.55	9.01
6,000	75 ± ½	88.5	53.5	7.0	46.5	14.37	4.08	10.29
7,000	75 ± ½	91.0	56.0	7.5	48.5	17.50	5.37	12.13
8,000	75 ± ½	93.5	59.5	8.0	51.5	21.20	6.42	14.78
9,000	75 ± ½	97.0	62.0	8.5	53.5	24.80	7.55	17.25
10,000	75 ± ½	100.5	64.5	9.0	55.5	28.60	9.53	19.07
			EMPTY BOBBIN					
5,000	75 ± ½	86.5	42.5	6.5	36.0	9.28	0.26	9.02
6,000	75 ± ½	88.5	46.0	7.0	39.0	12.05	0.37	11.68
7,000	75 ± ½	91.0	48.0	7.5	40.5	14.61	0.51	14.10
8,000	75 ± ½	93.5	50.0	8.0	42.0	17.30	0.66	16.64
9,000	75 ± ½	97.0	51.0	8.5	42.5	19.60	0.83	18.77
10,000	75 ± ½	100.5	56.0	9.0	47.0	24.20	1.03	23.17

TABLE III

102.6 S.U.V. (100°F.) Viscosity Oil

R.P.M. SPINDLE	AMBIENT AIR °F	OIL TEMPERATURE °F	θ_T °TWIST	θ_L °TWIST	θ_N °TWIST	W_T WATTS	W_A WATTS	W_B WATTS
1	2	3	4	5	6	7	8	9
			FULL BOBBIN					
5,000	75 ± ½	84.5	53.0	6.5	46.5	11.97	4.90	7.07
6,000	75 ± ½	86.5	61.5	7.0	54.5	16.83	8.10	8.73
7,000	75 ± ½	89.0	71.5	7.5	64.0	23.06	12.40	10.66
8,000	75 ± ½	91.0	78.5	8.0	70.5	29.03	16.70	12.33
9,000	75 ± ½	93.5	88.0	8.5	79.5	36.92	21.90	15.02
10,000	75 ± ½	97.0	102.0	9.0	93.0	47.90	29.90	18.00
			68% FULL BOBBIN					
5,000	75 ± ½	84.5	48.0	6.5	41.5	10.70	3.35	7.35
6,000	75 ± ½	86.5	55.0	7.0	48.0	14.84	5.93	8.91
7,000	75 ± ½	89.0	62.0	7.5	54.5	19.65	8.80	10.85
8,000	75 ± ½	91.0	68.0	8.0	60.0	24.79	11.56	13.23
9,000	75 ± ½	93.5	76.0	8.5	67.5	31.20	16.12	15.08
10,000	75 ± ½	97.0	85.0	9.0	76.0	39.12	22.10	17.02
			38.5% FULL BOBBIN					
5,000	75 ± ½	84.5	42.0	6.5	35.5	9.15	1.55	7.60
6,000	75 ± ½	86.5	47.0	7.0	40.0	12.35	4.08	8.27
7,000	75 ± ½	89.0	49.0	7.5	41.5	14.96	5.37	9.57
8,000	75 ± ½	91.0	43.5	8.0	45.5	18.74	6.42	12.32
9,000	75 ± ½	93.5	57.0	8.5	48.5	22.53	7.55	14.98
10,000	75 ± ½	97.0	61.5	9.0	52.5	27.05	9.53	17.52
			EMPTY BOBBIN					
5,000	75 ± ½	84.5	38.0	6.5	31.5	8.11	0.26	7.85
6,000	75 ± ½	86.5	38.0	7.0	31.0	9.58	0.37	9.21
7,000	75 ± ½	89.0	39.5	7.5	32.0	11.55	0.51	11.04
8,000	75 ± ½	91.0	41.0	8.0	33.0	13.59	0.66	12.93
9,000	75 ± ½	93.5	42.5	8.5	34.0	15.75	0.83	14.92
10,000	75 ± ½	97.0	44.0	9.0	35.0	18.09	1.03	17.06

TABLE IV

83.5 S.U.V. (100°F.) Viscosity Oil

R.P.M. SPINDLE	AMBIENT AIR °F	OIL TEMPERATURE °F	θ_T °TWIST	θ_L °TWIST	θ_N °TWIST	W_T WATTS	W_A WATTS	W_B WATTS
1	2	3	4	5	6	7	8	9
			FULL BOBBIN					
5,000	75 ± ½	83.0	50.0	6.5	43.5	11.20	4.90	6.30
6,000	75 ± ½	84.5	56.5	7.0	49.5	15.30	8.10	7.20
7,000	75 ± ½	86.0	67.0	7.5	49.5	21.42	12.40	9.02
8,000	75 ± ½	88.5	76.0	8.0	68.0	28.00	16.70	11.30
9,000	75 ± ½	91.5	87.0	8.5	78.5	35.95	21.90	14.05
10,000	75 ± ½	94.5	99.5	9.0	90.5	46.60	29.90	16.70
			68% FULL BOBBIN					
5,000	75 ± ½	83.0	43.0	6.5	36.5	9.40	3.35	6.05
6,000	75 ± ½	84.5	50.0	7.0	43.0	13.30	5.93	7.37
7,000	75 ± ½	86.0	55.5	7.5	48.0	17.30	8.80	8.50
8,000	75 ± ½	88.5	61.5	8.0	53.5	22.04	11.56	10.48
9,000	75 ± ½	91.5	71.0	8.5	62.5	29.00	16.12	12.88
10,000	75 ± ½	94.5	79.0	9.0	70.0	36.00	22.10	13.90
			38.5% FULL BOBBIN					
5,000	75 ± ½	83.0	38.5	6.5	32.0	8.24	1.55	6.79
6,000	75 ± ½	84.5	40.5	7.0	33.5	10.34	4.08	6.26
7,000	75 ± ½	86.0	44.5	7.5	37.0	13.34	5.37	7.97
8,000	75 ± ½	88.5	49.5	8.0	41.5	17.10	6.42	10.68
9,000	75 ± ½	91.5	52.5	8.5	44.0	20.40	7.55	12.85
10,000	75 ± ½	94.5	58.0	9.0	49.0	25.20	9.53	15.67
			EMPTY BOBBIN					
5,000	75 ± ½	83.0	34.0	6.5	27.5	7.08	0.26	6.82
6,000	75 ± ½	84.5	34.5	7.0	27.5	8.49	0.37	8.12
7,000	75 ± ½	86.0	36.0	7.5	28.5	10.28	0.51	9.77
8,000	75 ± ½	88.5	37.0	8.0	29.0	11.95	0.66	11.29
9,000	75 ± ½	91.5	38.0	8.5	29.5	13.68	0.83	12.05
10,000	75 ± ½	94.5	38.5	9.0	29.5	15.20	1.03	14.17

TABLE V

70.8 S.U.V. ($100^{o}F.$) Viscosity Oil

R.P.M. SPINDLE	AMBIENT AIR ^{o}F	OIL TEMPERATURE ^{o}F	θ_T ^{o}TWIST	θ_L ^{o}TWIST	θ_N ^{o}TWIST	W_T WATTS	W_A WATTS	W_B WATTS
1	2	3	4	5	6	7	8	9
			FULL BOBBIN					
5,000	$75 \pm \frac{1}{2}$	82.5	49.0	6.5	42.5	10.91	4.90	6.01
6,000	$75 \pm \frac{1}{2}$	84.0	54.5	7.0	47.5	14.60	8.10	6.50
7,000	$75 \pm \frac{1}{2}$	85.5	63.5	7.5	56.0	20.24	12.40	7.84
8,000	$75 \pm \frac{1}{2}$	87.5	72.5	8.0	64.5	26.53	16.70	9.83
9,000	$75 \pm \frac{1}{2}$	89.5	83.0	8.5	74.5	34.50	21.90	12.60
10,000	$75 \pm \frac{1}{2}$	93.0	95.5	9.0	86.5	44.50	29.90	14.40
			68% FULL BOBBIN					
5,000	$75 \pm \frac{1}{2}$	82.5	42.5	6.5	36.0	9.31	3.35	5.96
6,000	$75 \pm \frac{1}{2}$	84.0	47.5	7.0	40.5	12.52	5.93	6.59
7,000	$75 \pm \frac{1}{2}$	85.5	54.0	7.5	46.5	16.70	8.80	7.90
8,000	$75 \pm \frac{1}{2}$	87.5	60.0	8.0	52.0	21.47	11.56	9.91
9,000	$75 \pm \frac{1}{2}$	89.5	69.0	8.5	60.5	28.13	16.12	12.01
10,000	$75 \pm \frac{1}{2}$	93.0	80.0	9.0	71.0	36.60	22.10	14.50
			38.5% FULL BOBBIN					
5,000	$75 \pm \frac{1}{2}$	82.5	36.0	6.5	29.5	7.54	1.55	5.99
6,000	$75 \pm \frac{1}{2}$	84.0	41.5	7.0	34.5	10.59	4.08	6.51
7,000	$75 \pm \frac{1}{2}$	85.5	44.5	7.5	37.0	13.34	5.37	7.97
8,000	$75 \pm \frac{1}{2}$	87.5	47.5	8.0	39.5	16.28	6.42	9.86
9,000	$75 \pm \frac{1}{2}$	89.5	51.0	8.5	42.5	19.61	7.55	12.06
10,000	$75 \pm \frac{1}{2}$	93.0	56.0	9.0	47.0	24.20	9.53	14.67
			EMPTY BOBBIN					
5,000	$75 \pm \frac{1}{2}$	82.5	30.5	6.5	24.0	6.18	0.26	5.92
6,000	$75 \pm \frac{1}{2}$	84.0	31.5	7.0	24.5	7.57	0.37	7.20
7,000	$75 \pm \frac{1}{2}$	85.5	33.0	7.5	25.5	9.20	0.51	8.69
8,000	$75 \pm \frac{1}{2}$	87.5	33.5	8.0	25.5	10.50	0.66	9.84
9,000	$75 \pm \frac{1}{2}$	89.5	36.0	8.5	27.5	12.93	0.83	11.90
10,000	$75 \pm \frac{1}{2}$	93.0	37.5	9.0	28.5	14.70	1.03	13.67

TABLE VI

Air Friction Data

R.P.M. SPINDLE	θ_{d1} °TWIST	θ_{d2} °TWIST	θ_{d3} °TWIST	W_A °WATTS
1	2	3	4	5
		FULL BOBBIN		
5,000	18.0	1.0	19.0	4.90
6,000	25.0	1.2	26.2	8.10
7,000	33.0	1.4	34.4	12.40
8,000	39.0	1.6	40.6	16.70
9,000	45.5	1.8	47.3	21.90
10,000	56.0	2.0	58.0	29.90
		68% FULL BOBBIN		
5,000	12.0	1.0	13.0	3.35
6,000	18.0	1.2	19.2	5.93
7,000	23.0	1.4	24.4	8.80
8,000	26.5	1.6	28.1	11.56
9,000	33.0	1.8	34.8	16.12
10,000	41.0	2.0	43.0	22.10
		38.5% FULL BOBBIN		
5,000	6.0	1.0	7.0	1.80
6,000	12.0	1.2	13.2	4.08
7,000	13.5	1.4	14.9	5.37
8,000	14.0	1.6	15.6	6.42
9,000	14.5	1.8	16.3	7.55
10,000	16.5	2.0	18.5	9.53
		EMPTY BOBBIN		
5,000	0	1.0	1.0	0.26
6,000	0	1.2	1.2	0.37
7,000	0	1.4	1.4	0.51
8,000	0	1.6	1.6	0.66
9,000	0	1.8	1.8	0.83
10,000	0	2.0	2.0	1.03